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EVALUATION OF HOLLOWED (DRILLED) BALLS IN BALL BEARINGS AT DN VALUES TO 2 MILLION

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

ABSTRACT

Two 75-millimeter bore bearings having balls with a 50-percent weight reduction were operated at 500-lb. (2200N) thrust load at speeds to 28,000 rpm (2.1-million DN). The weight reduction was achieved by machining (drilling) a concentric hole through each ball. More than 100-hours running time was accumulated on one bearing, with 65-hours at DN values above 1.5 million. Bearing torque and outer race temperature data were obtained and compared with bearings having either spherically hollow or conventional solid balls. Both air-oil mist and oil jet lubrication were used.

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INTRODUCTION

Aircraft gas turbine engine rotor bearings currently operate at DN values (product of bearing bore diameter in millimeters and shaft speed in rpm) to approximately 2 million. It is estimated that engine designs of the next decade will require bearings to operate at DN values of 3 to 4 million. In this DN range, the reduction in bearing fatigue life due to the high centrifugal forces developed between the rolling elements and outer race becomes prohibitive.

To solve the problem of lower fatigue life in high-speed ball bearings, various methods of reducing ball mass to reduce the centrifugal force are being considered. Spherically hollow balls, which are fabricated by welding two hemispherical shells together, offer a possible solution. Experience to date with some of these has, however, been poor. Flexure failures which originated at the edges of the weld on the inside diameter have caused early spalling and even complete fracture of the ball. With spherically hollow balls, there may always be problems with stress raisers on the inside diameter because they cannot be inspected readily.

Another method of reducing ball mass is to machine a concentric hole through the ball. The amount of mass reduction equal to that of a thin-wall hollow ball can be achieved with several advantages. Two of these are the concentricity of the hole axis with the ball center (and thus the balance) can be maintained with high accuracy and the inside surface of the hole and the outer surface of the ball can be finished without surface irregularities or flaws such as may be present in the weld area of a spherically hollow ball.

The objectives of this investigation were: (1) to demonstrate the feasibility of the "hollowed" ball concept by operating a bearing at high rotative speeds, and (2) to compare the performance of bearings having "hollowed" balls with those having either spherically hollow balls or conventional solid balls over a range of shaft speeds.

Tests were conducted with 75 mm bore, angular contact, ball bearings operating with a 500 lb. thrust load at speeds from 10,000 to 28,000 rpm (0.75 to 2.1 million DN) using either air-oil mist or oil jet lubrication. Performance data from bearings with spherically hollow balls or conventional solid balls were compared with that obtained with the "hollowed" ball bearings over a range of operating conditions.

APPARATUS

Bearing Test Rig

A cutaway view of the bearing test apparatus is shown in figure 1. A variable-speed direct current motor drives the test shaft through a 14 to 1 gear speed increaser. The test bearing was thrust loaded through an externally pressurized gas thrust bearing, which also permitted measurement of the bearing torque. Bearing torque was measured by an unbonded strain-gage force transducer connected to the periphery of the test bearing housing. Bearing outer race temperature was measured with two iron-constantan thermocouples located as shown in figure 1. Lubricant was supplied to the test bearing in the form of air-oil mist or oil jet through the lubricant delivery tube. The lubricant was a super-refined napthenic mineral oil with a viscosity of 75 centistokes ($75 \times 10^{-6} m^2/sec$) at 100° F (311° K). Oil flows ranged from 0.01 to 0.07 lb/min (0.8×10^{-4} to 5.3×10^{-4} Kg/sec) with the air-oil mist system and was constant at 1.0 lb/min (76×10^{-4} Kg/sec) with the oil jet system.

A photograph of the test bearing used in the study is shown in figure 2. It is a 75-mm bore, angular contact, ball bearing with eleven 0.6875-inch (17.5-mm) diameter balls. The cage is of two-piece construction and is piloted on the outer-race land. One shoulder of the inner race is relieved to make the bearing separable. Bearing internal clearance measured approximately 0.0020-inches (0.05-mm).

Figure 3 is a section view of the test bearing showing details of the hollowed ball and the ball retaining axle. Each 0.6875-inch (17.5-mm) diameter ball has a 0.42-inch (10.7-mm) diameter concentric hole machined through it. This size hole results in a weight reduction of 50-percent from that of a solid ball. Each ball is retained in the cage by a 0.125-inch (3.1-mm) diameter axle which prevents the edge of the concentric hole from rubbing on the race groove. The axle is located at approximately the center of the ball pocket at the pitch diameter of the bearing.

RESULTS AND DISCUSSION

Bearing Tests

Two 75-millimeter bore bearings with hollowed balls were operated at 500-lb (2200N) thrust load over a range of shaft speeds from 10,000 to 28,000 rpm (0.75 to 2.1-million DN), using either air-oil mist or oil jet lubrication. Outer race temperature and torque data for these bearings were compared with similar data for bearings having conventional solid balls and for bearings having spherically hollow balls with 56-percent weight reduction.

Figure 4 (a) shows bearing outer-race temperatures as a function of shaft speed for the three ball design configurations using air-oil mist lubrication. The temperatures of the bearings with hollowed balls, 3SRH and 4SRH, compare favorably with the solid or spherically hollow ball bearings over the speed range.

With oil jet lubrication (figure 4 (b), the outer-race temperature of the two hollowed ball bearings was slightly lower than that of the solid ball bearing for the same range of speeds. These lower outer-race temperatures reflect lower surface temperatures of the hollowed balls and were probably due to two effects: (1) having lower ball mass to cool and (2) additional cooling surface available with the hole through the interior of the ball. Comparison of figures 4 (a) and 4 (b) shows that the bearing temperatures with oil-jet lubrication averaged about 70° F (294° K) cooler than with air-oil-mist lubrication over the same speed range.

These lower temperatures resulted from a 14-fold increase in oil flow and better cooling with oil jet lubrication.

Bearing torque as a function of shaft speed is compared in figure 5 for the three ball configurations. Using air-oil mist lubrication (figure 5 (a), the bearing with spherically hollow balls shows slightly lower torque values than the other bearings. These torque differences are not considered to be significant, however. Using oil jet lubrication (figure 5 (b), torque for the hollowed ball bearings, 3SRH and 4SRH showed about a 3-fold increase in torque with jet lubrication over that with air-oil mist. This increase probably resulted from oil churning within the bearings.

Bearing 3SRH had operated successfully for a total of 101-hours at speeds to 22,000 rpm (1.65×10^6 DN value). Accumulated running time at 20,000 rpm (1.5×10^6 DN) and above was 65 hours. Bearing 4SRH was successfully run to a maximum speed of 28,000 rpm (2.1×10^6 DN). Total run time to date on this bearing is 16 hours, with 5 hours above a DN value of 1.5-million.

Bearing Post-Test Inspection

Visual inspection of the hollowed ball bearings showed the races and cages were in good condition, without apparent wear or other damage. Photographs of one hollowed ball, axle, and retaining screw from bearings 3SRH and 4SRH are shown in figures 6 and 7, respectively. There was no evidence of abnormal ball-race tracking or skidding in either bearing 3SRH or 4SRH.

In figure 6, severe wear near the axle ends can be seen. Some wear on these axles was observed after bearing 3SRH had run only 7-hours. Figure 7, however, shows little wear on the axle of bearing 4SRH, after 16 hours running. It was assumed therefore, that the wear observed in figure 6 was initially caused by the sharp edge on the concentric hole through the ball. These edges on the balls of bearing 4SRH were chamfered which apparently lessened the wear on the axles, as can be seen in figure 7. Both axles were of stainless tubing.

SUMMARY OF RESULTS

Two 75-millimeter bore bearings having balls with a 50-percent weight reduction were operated at 500 lb. (2200N) thrust load at speeds to 28,000 rpm (2.1-million DN). Weight reduction was achieved by drilling a 0.42-inch (10.7 mm) diameter concentric hole through each ball. More than 100-hours running time was accumulated on one bearing. Of this total, 65-hours was obtained at DN values above 1.5-million.

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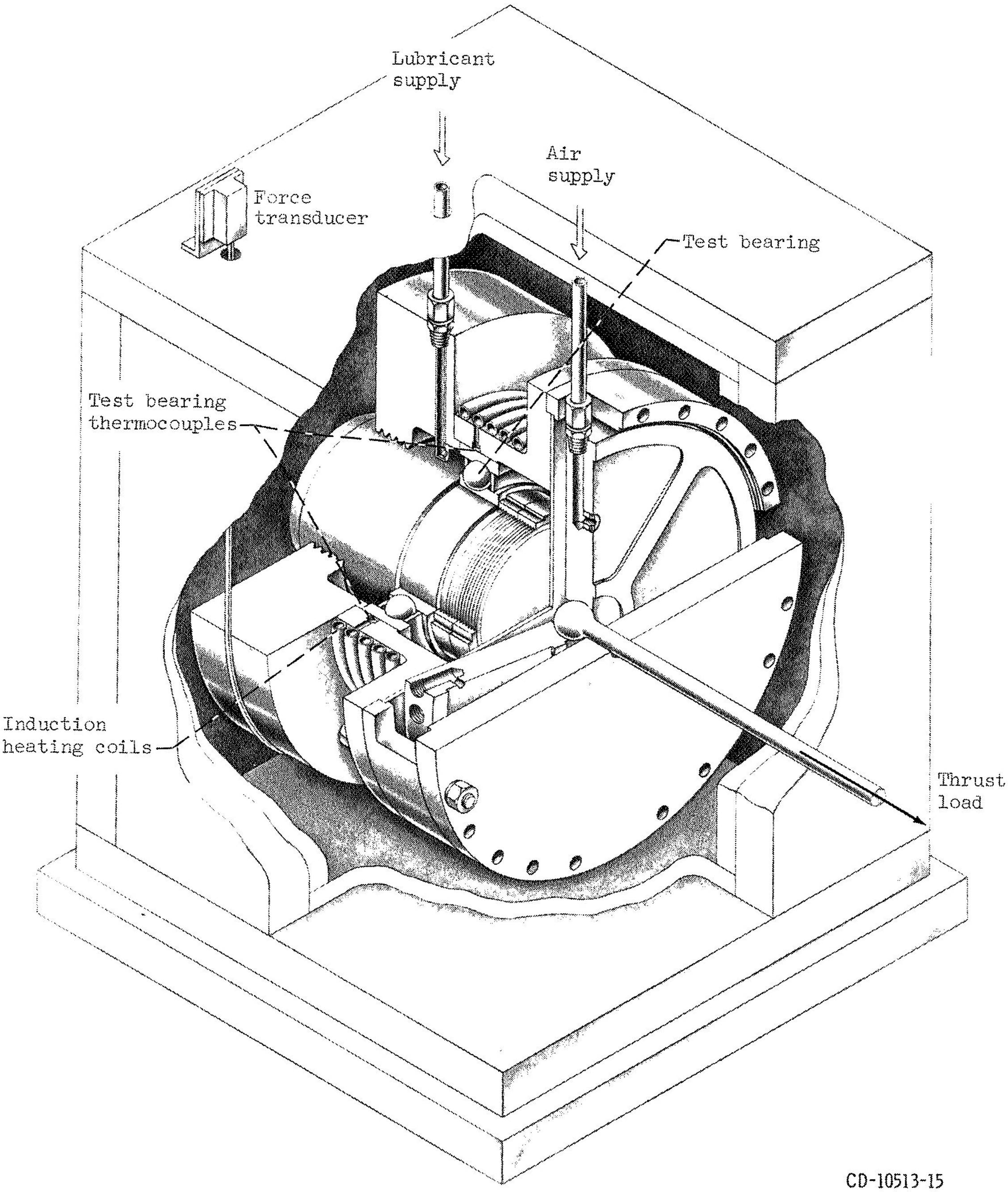

Figure 1. - Bearing test apparatus.

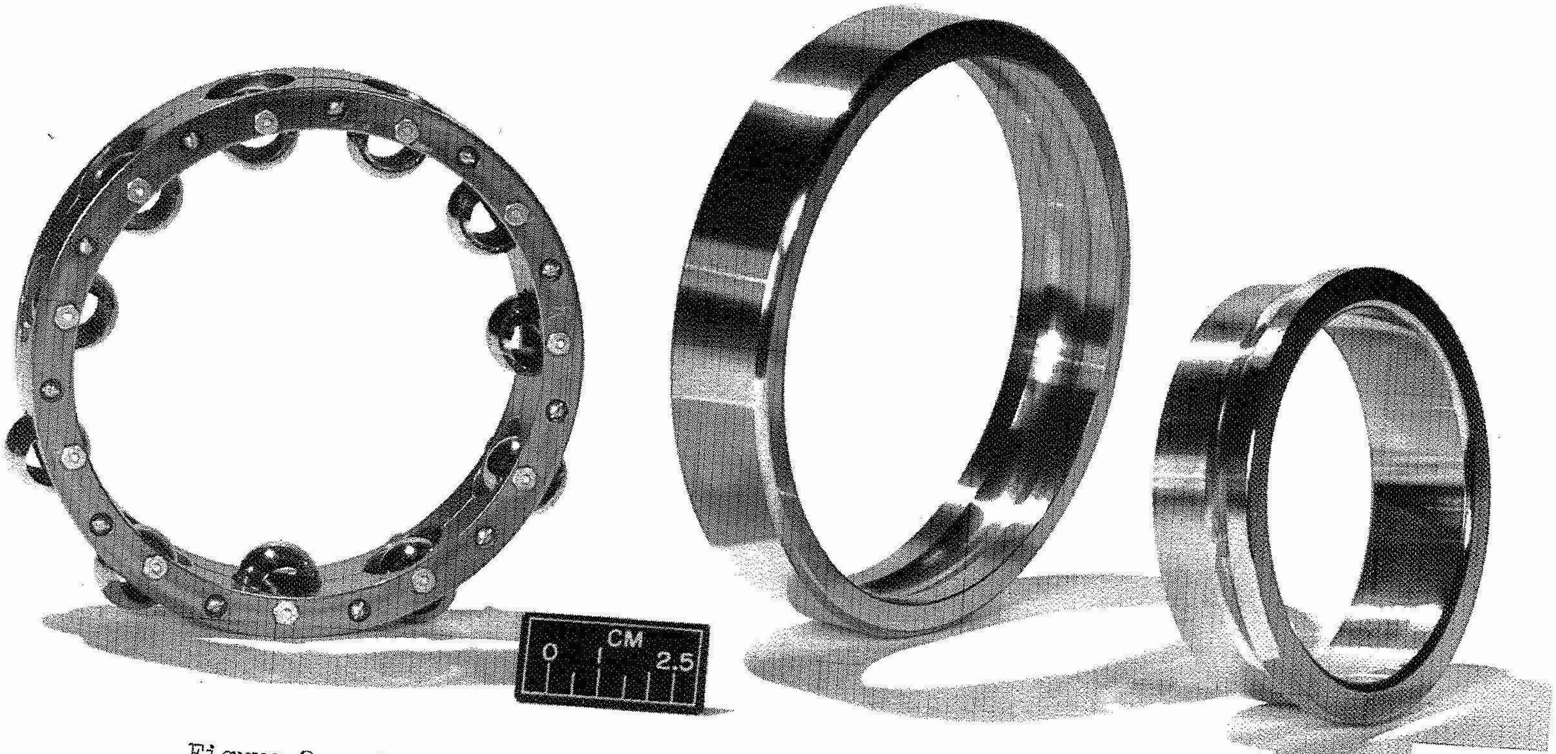

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Figure 2. - Test bearing - 75 mm bore size with machined hollow balls.

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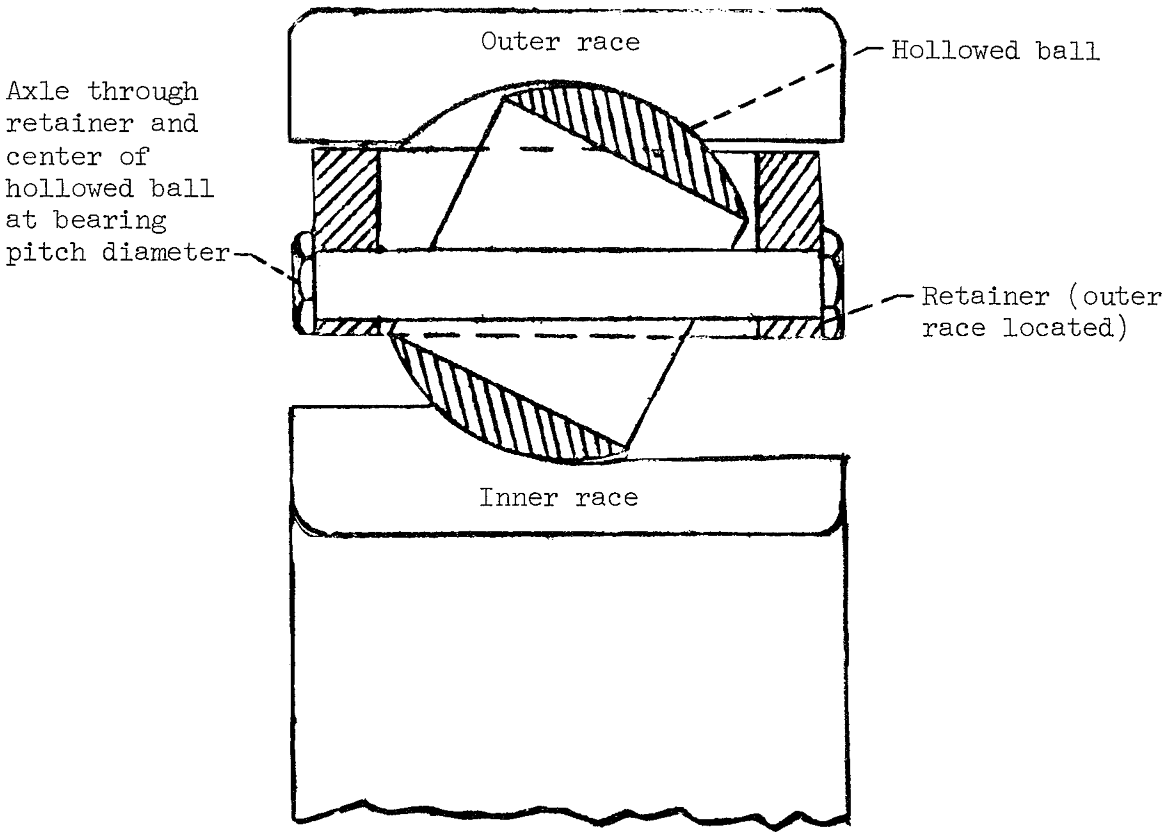

Figure 3. - Section view of test bearing showing hollowed ball and mounting detail.

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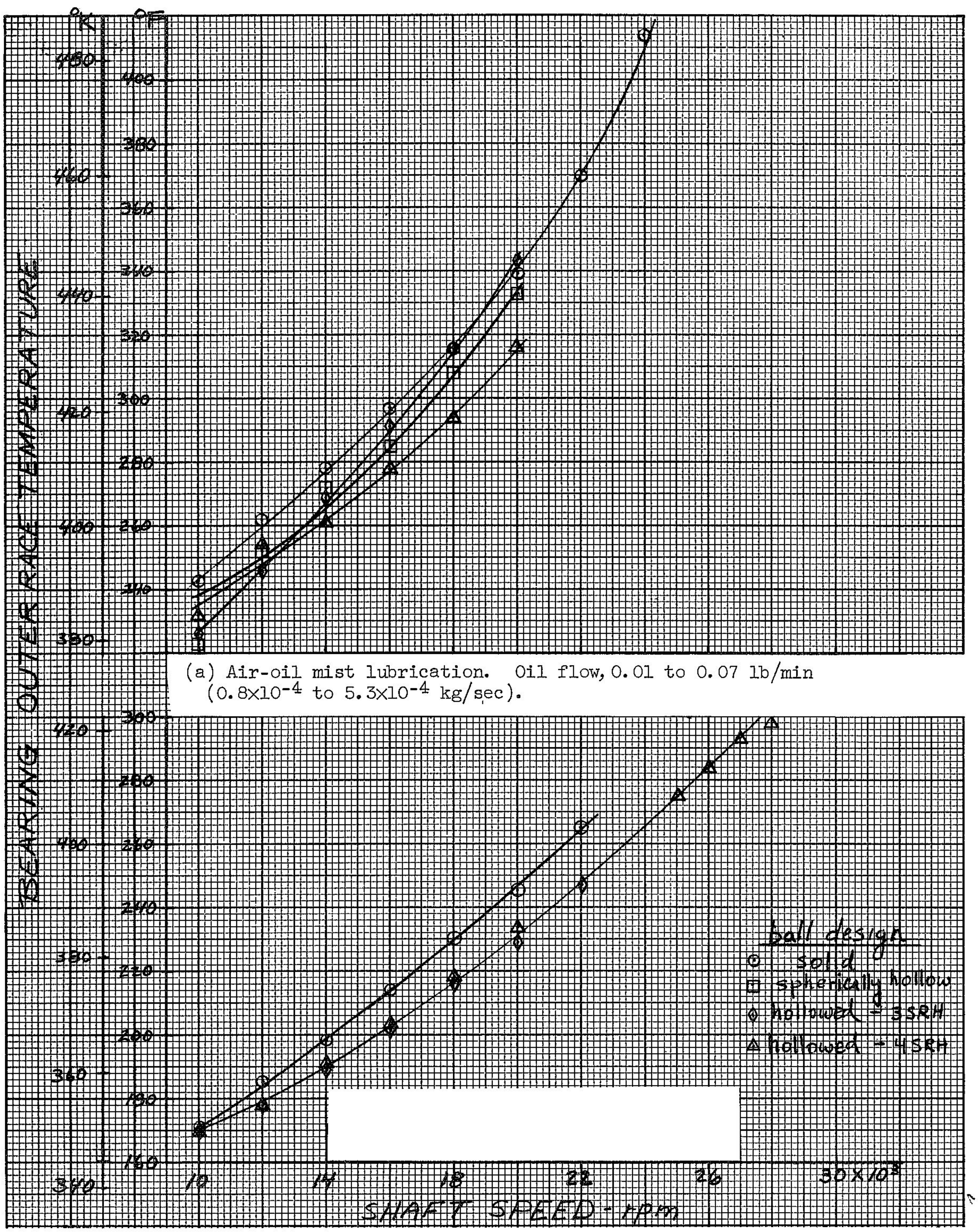

(a) Air-oil mist lubrication. Oil flow, 0.01 to 0.07 lb/min (0.8×10^{-4} to 5.3×10^{-4} kg/sec).

(b) Oil jet lubrication. Oil flow, 1.0 lb/min (7.6×10^{-3} kg/sec).

Figure 4. - Bearing outer race temperature as a function of shaft speed for bearings with solid, spherically hollow, and machined hollow balls. Thrust load, 500 lb (2200 N).

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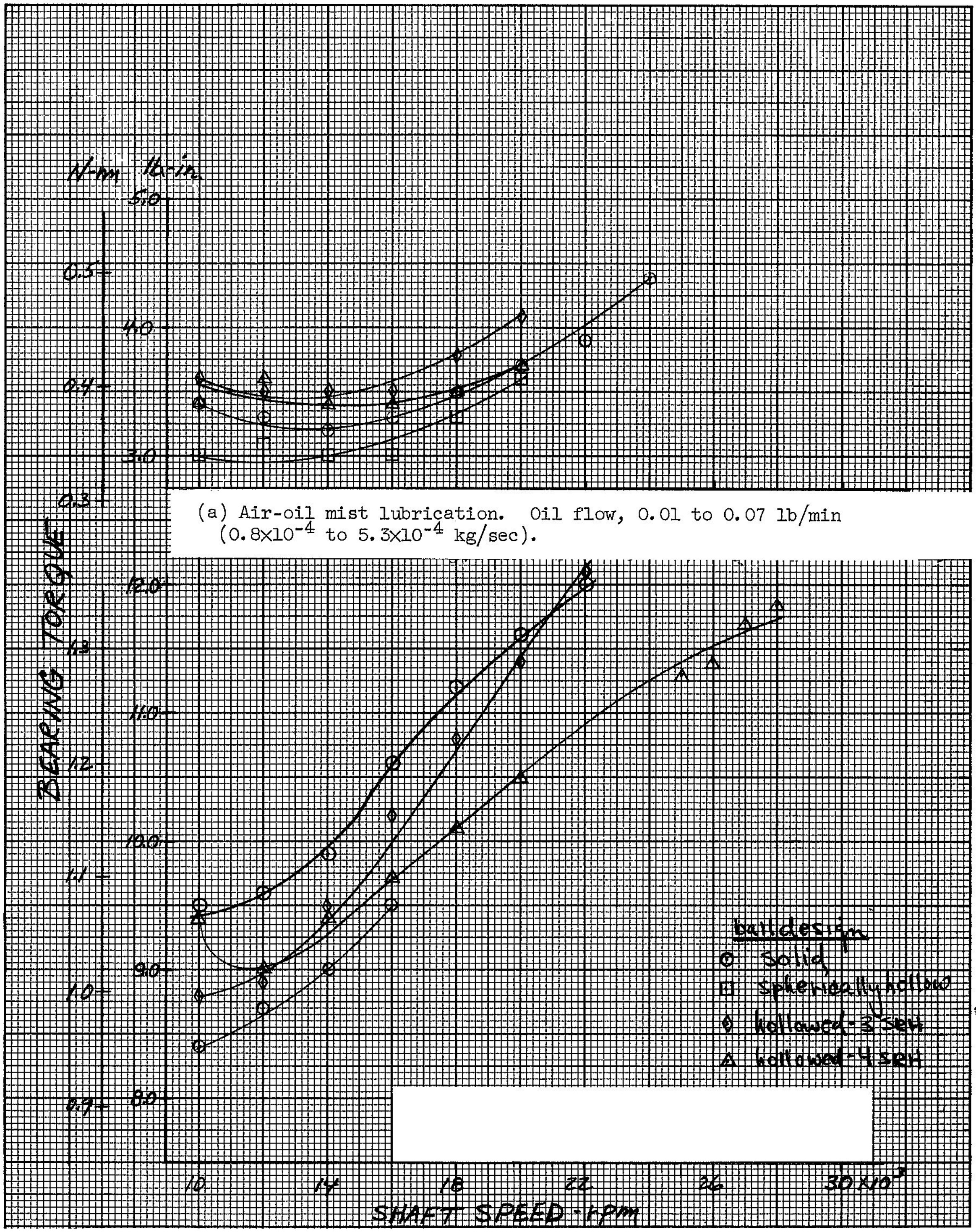

(b) Oil jet lubrication. Oil flow, 1.0 lb/min (7.6×10^{-3} kg/sec).

Figure 5. - Bearing torque as a function of shaft speed for bearings with solid, spherically hollow, machined hollow balls. Thrust load, 500 lb (2200 N).

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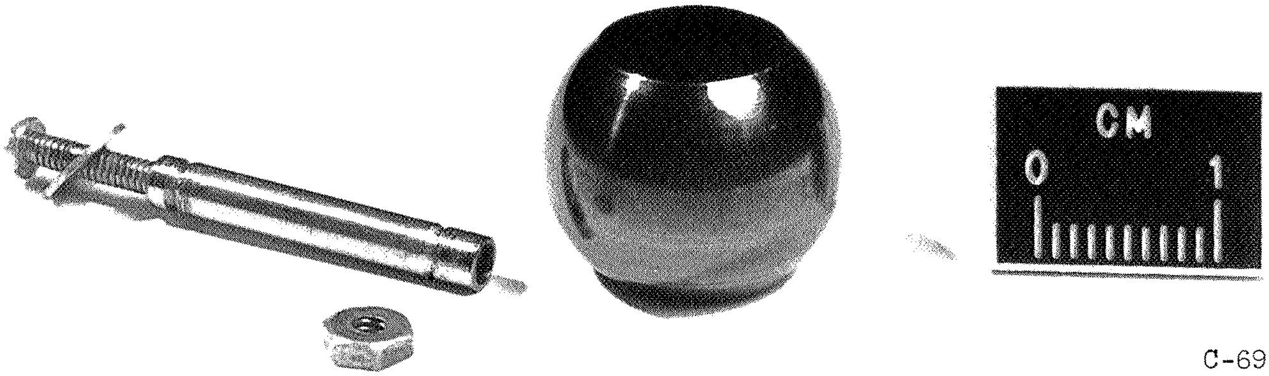

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Figure 6. - Hollowed ball and axle from bearing 3S-R-H after 101 hours of operation. Maximum speed, 22 000 rpm (1.65×10^{6} DN); thrust load, 500 lb (2200 N); running time above 1.5×10^{6} DN value, 65 hours.

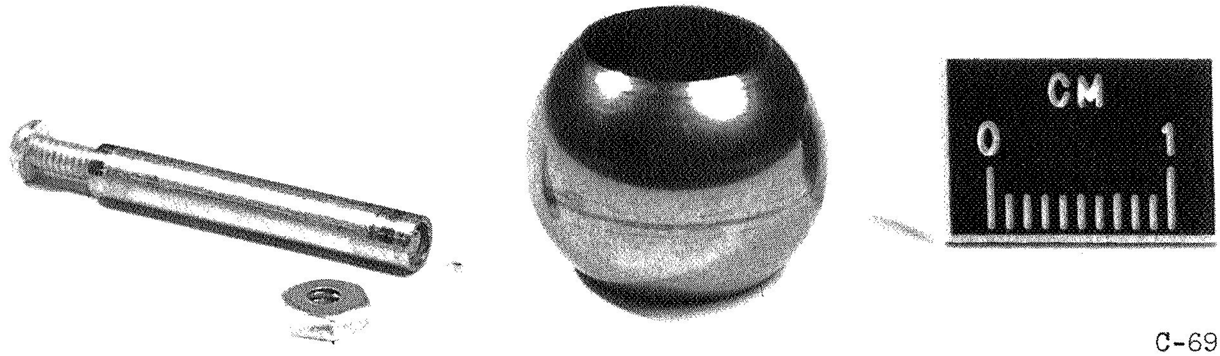

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Figure 7. - Hollowed ball and axle from bearing 4S-R-H after 16 hours of operation. Maximum speed, 28 000 rpm (2.1×10^{6} DN); thrust load, 500 lb (2200 N); running time above 1.5×10^{6} DN value, 5 hours.